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Optimization Of Passive Vehicle Suspension System By Genetic Algorithm

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Abstract

The design of suspension system has been one of the challenging tasks for engineers. The main function of any suspension system is to reduce or eliminate the road excitations transmitted to vehicle body. An effort has been made in this paper for a passive suspension system by using an optimization technique called Genetic algorithm to absorb vibrations as per ISO 2631-1: 1997 standards. The spring stiffness (ks), damping coefficient (cs), sprung mass (ms), unsprung mass (mu), tyre stiffness (kt) are optimized in such a way that ride comfort is increased .The quarter car and driver seat with the driver's body are simply modeled together as four degree of freedom (DOF) system by using SIMULINK for analyzing the ride comfort.

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Keywords: Quarter Car Model, Optimization, Ride Comfort, Genetic Algorithm, Simulink.

1. Introduction

Passenger comfort is very important parameter in design and manufacturing of automobiles. Suspension system carries weight of vehicle structure, driver and passenger and it absorbs the vibrations passing to the vehicle body from road. These vibrations are mainly due to irregularities in road surface, aerodynamic forces and non-uniform wheel assembly and out of these surface irregularities are predominant.

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Ride comfort (RC) is defined as the condition where the passenger experiences no or minimum shocks that are transferred by the road surface. So minimum is the sprung mass acceleration maximum is the RC. Ride Comfort is defined by the ISO 2631-1-1997. There are many harmful effects on body due to vehicle vibrations. Kjellberg [1] has stated that the abdominal walls causes the hyperventilation, disorders of the back like back pain, osteoarthritis, slipping of disc etc. are commonly associated diseases with exposure to vibration. Ride comfort is measured in terms of sprung mass acceleration.

Many research activities have been carried out in the field of optimization to improve ride comfort. Griffin et al. [2] has performed a detailed experimental work to determine the effects of road excitation on ride comfort. Many optimization methods which has been in use are gradient based methods, rotating square evolutionary operation (ROVOP), genetic algorithm and sequential search method. Anil Shirahatti et al. [3] have used Genetic Algorithm (GA) to minimize number of objectives like sprung mass acceleration, jerk and road holding and the results were compared in a Simulink model. Tewari et al. [4] and Gundogdu [5] determined a set of optimized parameters by using GA for a 4-DOF quarter car seat and suspension model to achieve the best performance for the driver's seat. As per ISO 2631-1: 1997 [6], passenger comfort depends on root mean square (RMS) value of acceleration and the frequency of vibrations acting on body. The ability of the tire of vehicle to remain in contact with the road surface is known as road holding (RH).

There are many harmful effects on body due to vehicle vibrations. Mehrdad N. Khajavihas et al. [7] developed a full car 8 degrees of freedom model and done research on three criteria, namely ride comfort, handling and working space and has been adopted as objective functions. G. Verros et al. [8] has presented methodology for optimizing the suspension damping and stiffness parameters of nonlinear quarter-car models subjected to random excitation.

In this paper a 4-DOF quarter car model with Thorax and Pelvis is used to analyse five suspension parameters namely spring stiffness (ks), damping coefficient (cs), sprung mass (ms), unsprung mass (mu), tyre stiffness (kt) by Genetic Algorithm. Also an objective function is derived with above said five variables with a Modal Analysis theory and using Symbolic Math toolbox of MATLAB. In the course of the analysis, assumption is made that springs and dashpots used in this model are weightless and have linear time-independent characteristics.

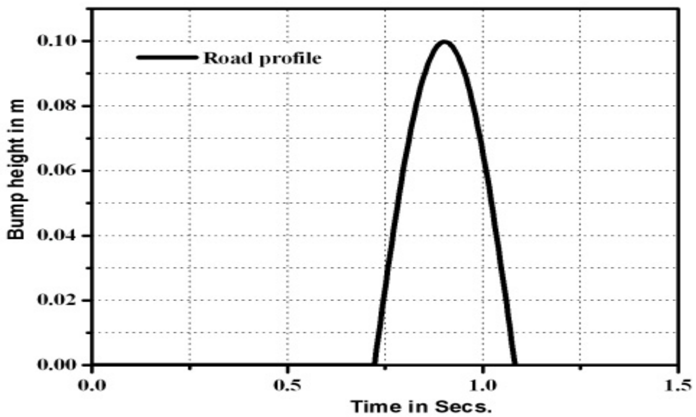

Fig. 1. Road profile of height 0.1m

$$
\begin{aligned}
zr &= 0 \quad \text{when } t < \frac{d}{v} \\
&= h \times \sin\left[\frac{\Pi \times v}{l} \times \left(t - \frac{d}{v}\right)\right] \text{ for } \frac{d}{v} \le t \le \frac{d+l}{v} \\
&= 0 \quad \text{when } t > \frac{d+l}{v}
\end{aligned} \tag{1}
$$

In the Eq. 1 ‘v’ is the velocity in m/s, ‘d’ is the distance between front and rear axle in meter and ‘t’ is the time lag between the crossing of front and rear wheels across the bump in sec. Figure 1, shows road profile which is half sinusoidal wave of height 0.1 m. This profile is created using MATLAB coding and then given as input to SIMULINK model [9].

2. Derivation of objective function by Modal Analysis

The objective of this paper is to improve the ride comfort as per standard ISO 2631-1:1997, because of the conflicting nature of RC and RH and finding corresponding optimized parameters of spring stiffness (ks), damping coefficient (cs), sprung mass (ms), unsprung mass (mu) and tyre stiffness (kt) using an optimization technique called Genetic Algorithm. The quarter car model is used in the analysis as it is a basic and simple model to analyze only vertical motion.

For the optimization, there is a need to formulate the objective functions for the quarter car model with variables ks, cs, ms, mu and kt. The objective function governing the sprung mass acceleration i.e. RC has been derived using modal analysis concept [10].

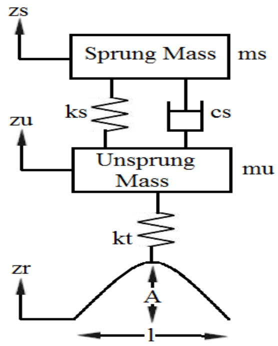

Fig. 2. Quarter car model

As per Newton’s 2nd law of motion Eq. (2) and (3) are written based on F.B.D of Quarter Car Model in Figure 2 corresponding to sprung and unsprung mass respectively.

$$ms \times \ddot{z}s + cs \times (\dot{z}s - \dot{z}u) + ks \times (zs - zu) + ms \times g = 0 \tag{2}$$

$$mu \times \ddot{z}u - cs \times (\dot{z}s - \dot{z}u) - ks \times (zs - zu) + kt(zu - zr) + mu \times g = 0 \tag{3}$$

$$RC = \frac{ms\left(\frac{ks}{ms} - \frac{\sigma_2}{2ms\,mu}\right)\left(\frac{\gamma1 + \gamma2}{\gamma3}\right) - \frac{cs^2(\gamma4 - \gamma5)\sigma_2{}^2}{\gamma6} + \frac{cs(\gamma7 + \gamma8)\sigma_2}{\gamma9} - \gamma10}{ks} + \frac{ms\left(\frac{ks}{ms} - \frac{\sigma_5}{2ms\,mu}\right)\left(\frac{\gamma1 + \gamma12}{\gamma3}\right) - \frac{cs^2(\gamma4 - \gamma13)\sigma_5{}^2}{\gamma6} + \frac{cs(I + \gamma14)\sigma_5}{\gamma15} - \gamma16}{ks} \tag{4}$$

$$RH = -\frac{\frac{\gamma 17\times(\gamma 18-\gamma 19)}{\gamma 21}-\gamma 22}{\gamma 24}-\frac{\frac{\gamma 25\times(\gamma 26-\gamma 27)}{\gamma 28}-\gamma 29}{\gamma 31}-\gamma 23-\gamma 8-A\sin(\omega t) \qquad (5)$$

The MATLAB coding generates sprung mass displacement, which is carried over to the MATLAB symbolic toolbox known as MUPAD for finding out acceleration i.e. RC by performing double derivative of displacement in terms of five variables ks, cs, ms, mu and kt, as in Eq. (4). Apart from RC, the equation of RH, relative displacement of unsprung mass and road input is also derived from coding as shown in Eq. (5).

3. GENETIC ALGORITHM (GA)

Genetic algorithms are based on the principles of natural genetics and natural selection. The basic elements of natural genetics are performed by three operations as Reproduction, Crossover, and Mutation are used in the genetic search procedure [11].

3.1.GA MATLAB Toolbox

All three operations are coded in MATLAB toolbox named Global Optimization [12]. For optimization of RC, Eq. (4) and of RH, Eq. (5) is entered in .m file.

3.2. Genetic Algorithm for Multiobjective

As RC and RH are conflicting parameters a multi objective Genetic Algorithm optimization technique is performed in this paper. Pareto chart in Figure 3 contains RC values on x axis and RH values on y axis. There are total 27 optimized points on the chart. The points having lower acceleration values i.e. the points which lie on left uppermost corners are indicating maximum RC and similarly, points which lie on right bottommost corners are indicating maximum RH. So designer has to decide which parameter to be preferred more as per utility of the vehicle [13].

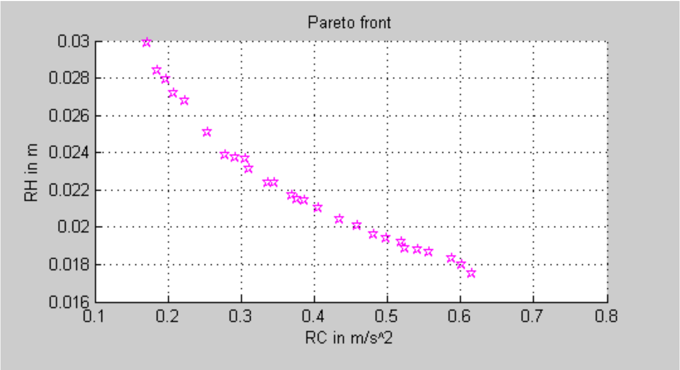

Fig. 3. Pareto Front

Among these 27 optimized points, in this study, RC has been focused with an aim that RH should not decrease more. From the Pareto front maximized RC with compromised RH value parameters setting as on Table 1 are selected.

Table 1. Values Optimized by GA for maximized RC setting

ks(N/m)	cs(Ns/m)	ms(kg)	kt(N/m)	mu(kg)	RC(m/s^2)	RH(m)
19290.86	1161.928	289.081	101428.1	29.92685	0.4048094	0.02101

4. Mathematical model

The quarter car with driver seat and the driver's body are simply modeled together as 4-DOF damped spring mass system for analysing the RC as shown in Figure 4. The tire has been replaced by equivalent spring stiffness, kt neglecting the damping. The wheel assembly, sprung mass, driver seat and passenger seat are modeled by linear springs in parallel with dampers. For this model four Eqs. (6 - 9) of motion are generated by using Newton's 2nd law of motion and free body diagram as shown in Figure 5. The displacements of all masses and road profile are as shown in Figure 4. The Table 2 shows the nomenclature of all parameters and corresponding values [10].

Table 2. Nomenclature of parameters of Quarter Car with Thorax and pelvis Model [5], [14]

Description	Parameters	Values
Mass of driver head (kg)	mh	20
Mass of driver seat with lower body (kg)	mc	45
Stiffness of driver body (N/m)	kb	45000
Stiffness of driver seat cushion (N/m)	kc	20000
Damping coefficient of driver body (Ns/m)	cb	1360
Damping coefficient of driver seat cushion (Ns/m)	cc	1650
Mass of sprung mass (kgs)	ms	241.65
Mass of unsprung mass (kgs)	mu	21.46
Suspension spring stiffness (N/m)	ks	21027.63
Suspension damping coefficient (Ns/m)	cs	1183.2
Tire stiffness (N/m)	kt	102017.2
Bump height (m)	A	0.1
Bump width (m)	ℓ	1

In Figure 4 zs is the displacement of sprung mass (ms), zu is the displacement of unsprung mass (mu), zr is the road input, zh is the displacement of head and zc is the displacement of the driver seat. All displacements are in meter. From Figure 5 and using Newton's second law four equations of motion are obtained Eqs. (6) – (9).

Sprung mass

$$\mathrm{ms}\times\ddot{\mathrm{z}}\mathrm{s} = \mathrm{cc}\times(\dot{\mathrm{z}}\mathrm{c}-\dot{\mathrm{z}}\mathrm{s})+\mathrm{kc}\times(\mathrm{zc}-\mathrm{zs})-\mathrm{cs}\times(\dot{\mathrm{z}}\mathrm{s}-\dot{\mathrm{z}}\mathrm{u})-\mathrm{ks}\times(\mathrm{zs}-\mathrm{zu})-\mathrm{ms}\times\mathrm{g} \quad (6)$$

Unsprung mass

$$\mathrm{mu}\times\ddot{\mathrm{z}}\mathrm{u} = \mathrm{cs}\times(\dot{\mathrm{z}}\mathrm{s}-\dot{\mathrm{z}}\mathrm{u})+\mathrm{ks}\times(\mathrm{zs}-\mathrm{zu})-\mathrm{kt}\times(\mathrm{zu}-\mathrm{zr})-\mathrm{mu}\times\mathrm{g} \quad (7)$$

Seat

$$\mathrm{mc}\times\ddot{\mathrm{z}}\mathrm{c} = -\mathrm{kc}\times(\mathrm{zc}-\mathrm{zs})-\mathrm{cc}\times(\dot{\mathrm{z}}\mathrm{c}-\dot{\mathrm{z}}\mathrm{s})+\mathrm{kb}\times(\mathrm{zh}-\mathrm{zc})+\mathrm{cb}\times(\dot{\mathrm{z}}\mathrm{h}-\dot{\mathrm{z}}\mathrm{c})-\mathrm{mc}\times\mathrm{g} \quad (8)$$

Head

$$\mathrm{mh}\times\ddot{\mathrm{z}}\mathrm{h} = -\mathrm{cb}\times(\dot{\mathrm{z}}\mathrm{h}-\dot{\mathrm{z}}\mathrm{c})-\mathrm{kb}\times(\mathrm{zh}-\mathrm{zc})-\mathrm{mh}\times\mathrm{g} \quad (9)$$

5. SIMULINK Model

The Simulink model of quarter car consisting of driver seat with driver body has been created by using Eqs. (6) – (9) of motion as shown in Figure 6. The road input is given as shown in Figure 1.

6. Results and Discussions

Comparative analysis of the before and after optimized settings is performed on a SIMULINK model of a quarter car with driver model as shown in Figure 6 with road input of single half sine bump as in Figure 1. The effects of optimization on sprung mass acceleration (zsdd) i.e. RC, road holding (RH) and driver head acceleration (zhdd) are studied and compared with the after optimized setting mentioned in Table 1 and before optimization setting as per Table 2. The rms values and the corresponding percentage changes in these parameters are shown in Table 3.

The graph of sprung mass acceleration is shown in Figure 7 shows that decay in corresponding amplitudes after optimization are more than before optimization which indicates improvement in ride comfort. This implies that for the same road input, the disturbance experienced by the sprung mass dies out faster for the optimized setting. On calculating rms values for both the graphs, it is found that, there is decrease in zsdd by 13.3% and corresponding improvement in ride comfort.

Table 3. Percentage changes in before and after optimization

Conditions	zsdd(m/s^2)	RH (m)	zhdd(m/s^2)
Before optimization	0.2759	0.0389	0.39797
After optimization	0.2392	0.0401	0.39926
% change	13.3	-3.08	-0.324

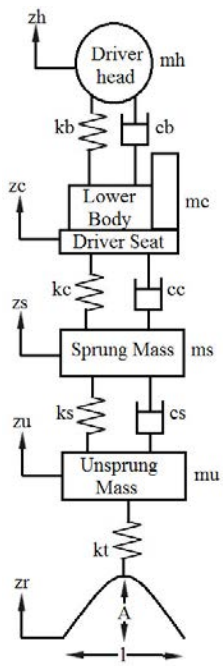

Fig. 4. Quarter car with Thorax and Pelvis

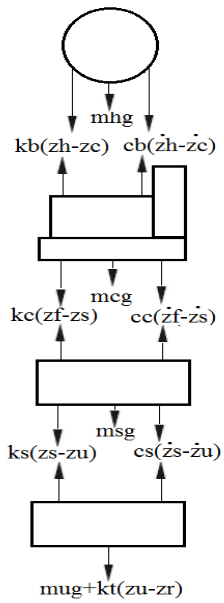

Fig. 5. Free body diagram of Quarter car with Thorax and pelvis

On the other hand, there is a decrease in rms value of road holding by 3.08% as seen in Figure 8. This is in accordance with the behaviour stated in literatures as maximizing only RC has an adverse effect on RH which is neglected in the optimization. The deviation of the mean value of relative displacement after stabilizing is due to the static compression of the tire. The change in the mean value of RH after stabilizing is attributed to the change in the sprung mass before and after optimizing.

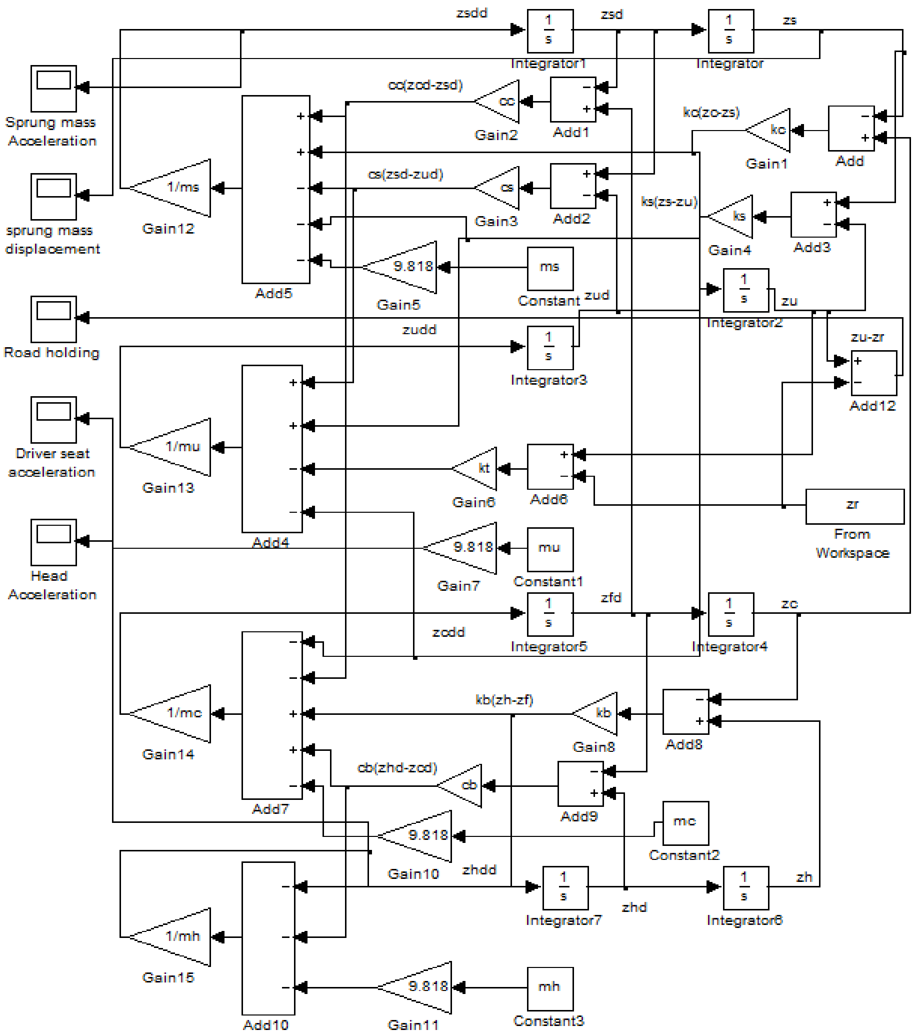

Fig. 6. Simulink Model of Quarter Car with Thorax and Pelvis

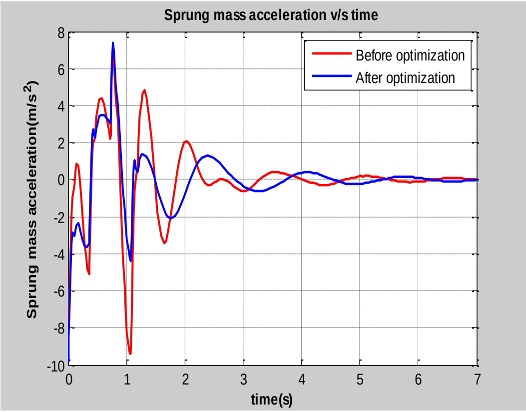

Fig. 7.Sprung mass acceleration v/s time for maximum RC

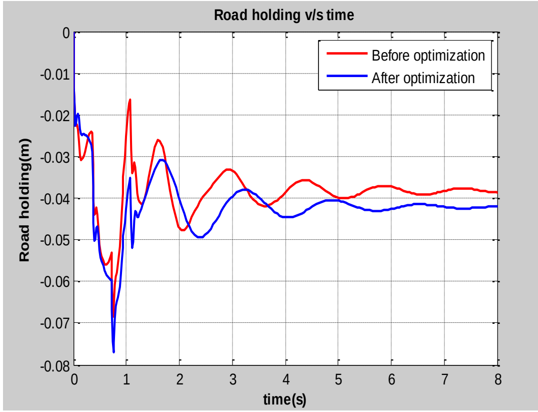

Fig. 8. Road holding v/s time for maximum RC

Figure 9 shows that the rate of decay of amplitude of driver head acceleration is greater in the optimized setting. It is clearly seen that stability is achieved in lesser cycles which results in increase in comfort of the driver. But the initial magnitude of acceleration is more in the optimized case. This result in greater rms acceleration of the optimized setting when compared to the non-optimized one. On the whole, the driver head acceleration value increases by 0.324%.

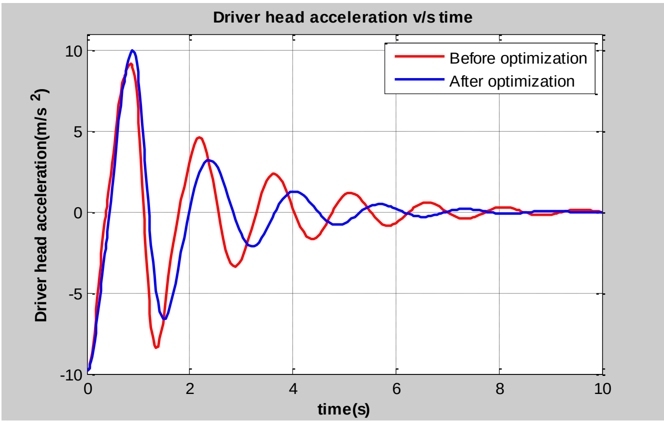

Fig 9. Driver head acceleration v/s time for maximum RC

It can be concluded that as per literature if RC is optimized the RH value deteriorates significantly but performing multi-objective optimization RC in improved more compared to the non-optimized setting with small amount of decrease on RH value.

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